View metadata, citation and similar papers at core.ac.uk

brought to you by CORE

provided by Purdue E-Pubs

Purdue University
Purdue e-Pubs

International Compressor Engineering Conference School of Mechanical Engineering

2000

A Complete Analysis of Dynamic Behaviour of Hermetic Compressor Cavity to Improve the Muffler Design

M. D. Libera
Electrolux Compressors

A. Faraon
Electrolux Compressors

A. Solari
Electrolux Compressors

Follow this and additional works at: https://docs.lib.purdue.edu/icec

Libera, M. D.; Faraon, A.; and Solari, A., "A Complete Analysis of Dynamic Behaviour of Hermetic Compressor Cavity to Improve the Muffler Design" (2000). *International Compressor Engineering Conference.* Paper 1448.
https://docs.lib.purdue.edu/icec/1448

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

A COMPLETE ANALYSIS OF DYNAMIC BEHAVIOUR OF HERMETIC COMPRESSOR CAVITY TO IMPROVE THE MUFFLER DESIGN.

Mirko Della Libera Alessio Faraon Alessandro Solari

Electrolux Compressors, Zanussi Elettromeccanica, R&D/Italy
Via Consorziale 13, 33170 Pordenone, Italy
Tel.0039-434-393489 Fax.0039-434-393313
e-mail: alessio.faraon@notes.electrolux.it

ABSTRACT

This paper focuses on dynamic behaviour of acoustic cavity of hermetic reciprocating compressor to develop a more refined method of muffler design. Study of compressor cavity modes is approached by creating a numerical model with experimental validation included. Some additional information has been collected to obtain a more realistic and correct muffler design considering cavity behaviour, this has been developed by means of a numerical model and by analysing experimental measurements.

1.INTRODUCTION

The importance of suction muffler from the point of view of thermodynamic efficiency and noise emission of reciprocating compressor is well known. This paper describes the activity performed for a refinement of suction muffler design from the acoustical point of view. Muffler design from the acoustical point of view used to be performed using Transmission Loss index only. TL index is a very important index at design stage, but it is not sufficient to obtain a complete description of acoustic attenuation introduced by the muffler. It is necessary to take working conditions into account to obtain a more realistic simulation. A complete numerical model of the muffler-cavity system has been prepared to perform a more accurate design. The whole activity has been based on a compressor with a refrigerant capacity 130 Kcal/h working in R134a fluid, this compressor model was characterised by a noise emission problem connected to the suction part.

2.NUMERICAL MODEL

Numerical model representing the airborne complete system has been analysed with SYSNOISE code [1], FEM methodology has been chosen for the first part of the analysis. System has been modelled in a very refined way in order to get a precise comparison with experimental results, internal parts of compressor like springs, stator feet, cluster and oil level... has been modelled. The highest frequency correctly calculated for the model was 2600Hz with the FEM approach (obtained using 25.010 nodes and 104.388 elements). In this model compressor shell was considered perfectly rigid without any fluid-structure coupling. Speed of sound and gas density has been chosen through measurements of gas temperature in various points of the internal cavity. Speed of sound differences have been taken into account by using of the FEM approach.

3.EXPERIMENTAL MODEL

Frequency response and operational deflections shape (Running Modes) technique have been chosen for the validation of the numerical model of the cavity. Muffler validation techniques are quite well established [2]. While determination of cavity resonance frequency is well known[3], a complete validation should be obtained through cavity modes individuation, which is not an easy task when testing at operating conditions. Visual representation of operational deflection shapes of acoustic pressure inside cavity has been obtained, frequency of modes and neighbouring harmonic peaks has been considered. Lms Running mode Analysis [4] has been used to process experimental measurements. Lms Fourier and Time monitors has been used to collect experimental data. The experimental set-up consisted of :

- 29 sound pressure measuring points on the shell internal surface
- measurement of sound pressure level at suction gas inlet
- a signal that indicates the temporal position of the top dead end of the piston (used to trigger measurements)
- two thermos-couples for detecting gas temperature inside shell.

Two 4182 B&K microphones have been used for measurements. Calibration has been accurately checked, length of microphone connecting pipe has been taken into account at calibration level. One microphone measured sound pressure level at muffler gas inlet, the other roving on the 29 points at internal shell surface. Top-dead-end signal was measured simultaneously to pressure measurements. All signals has been acquired as time-data streams to ease post-processing. Two approaches have been applied: Transmissibility functions (1), that takes into account variation of working conditions and Crosspowers (2), that improves phase relationship together with the top-dead-end.

$$X_i(\omega)=T_{ij}(\omega)*\sqrt{G_{ij}(\omega)} \qquad (1) \qquad X_i(\omega)=\frac{G_{ij}(\omega)}{\sqrt{G_{jj}(\omega)}} \qquad (2)$$

where: $T_{ij}(\omega)$ Transmissibility function $G_{ij}(\omega)$ Crosspower spectra $G_{jj}(\omega)$ Autopower spectra $X_i(\omega)$ Output signals

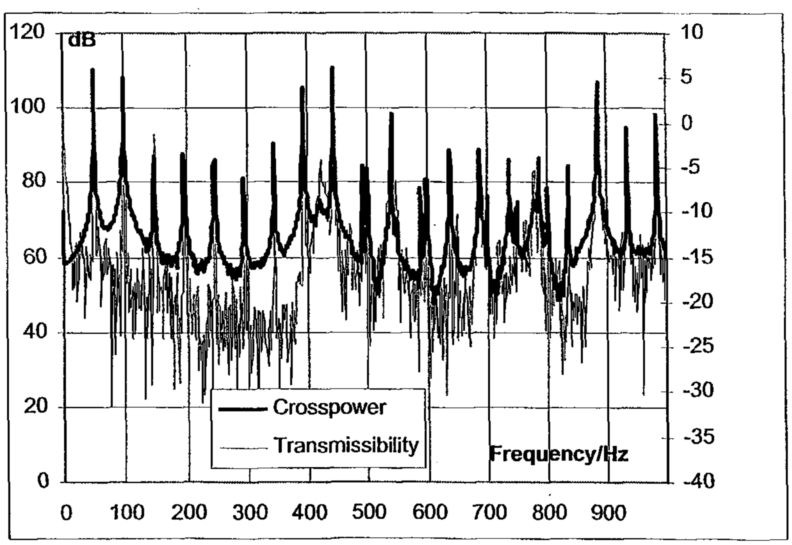

Fig.1 Crosspower and transmissibility methods comparison.

Considering R134a fluid, frequency limit with this set-up was at about 800Hz, on this way the first 6 modes of the cavity have been precisely described.

4.RESULTS:

As can be seen from the following table numerical and experimental data is very consistent in the considered frequency range.

Mode nr.	Experimental	Numerical
1	421	411
2	470	474
3	540	522
4	589	598
5	686	666
6	778	748

Tab.1 Acoustic Modes of the cavity Numerical-Experimental comparison.

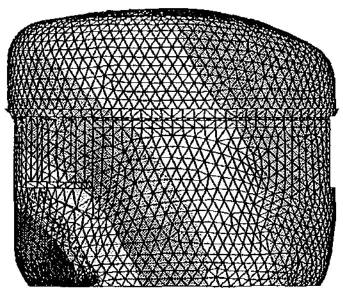

Fig.2 First computed mode

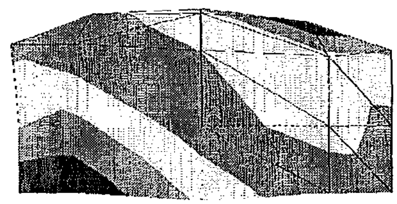

Fig.3 First experimental mode

Validation on the remaining frequency range has been obtained by considering frequency response of the system to a pressure field at muffler gas inlet, with a pressure corresponding to the measured acoustic pressure on the same point. Numerical resulting pressure has been compared to experimental acoustic pressure on the some points of the cavity.Results are shown in Figure.4 and 5 (2 points of the cavity), there is quite a good agreement between numerical evaluations and experimental measurements. A damping factor estimated from measurements has been introduced at numerical evaluation level.

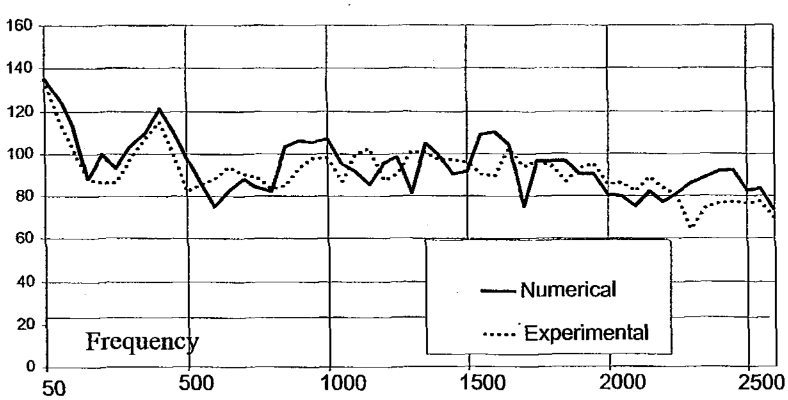

Fig.4 Sound pressure level inside cavity Point2.

5.MODEL IMPROVEMENT

A validated model of the muffler-cavity system gives a more realistic indication at design stage that takes into account operating conditions, which the Transmission Loss "aseptic" index does not give the opportunity of covering. The first step to simulate the effective working conditions of muffler used to consider a system radiating in an infinite space, with an impedance which could be represented by the radiation impedance [5], known as:

$$Z_r = \rho c S\left(\frac{1}{4}(ka)^2 + j0.61ka\right)$$ considering an unflanged termination (3)

where:

$k = wave\,number \quad a = tube\,diameter \quad S = tube\,surface \quad c = speed\ of\ sound$

This hypothesis did not consider dynamic behaviour of the cavity and the resulting effect on muffler acoustic performance, therefore numerical model has to consider muffler plus cavity system. Difference between the two approaches can be seen in Fig.6. which compares the curve of evaluated theoretical radiation impedance with (3) and impedance at muffler gas inlet numerically evaluated. As can be seen impedance of the complete system follows the trend of the radiation impedance, most appreciable differences are at low frequency in the range of cavity resonance. This fact can change dynamic behaviour of muffler.

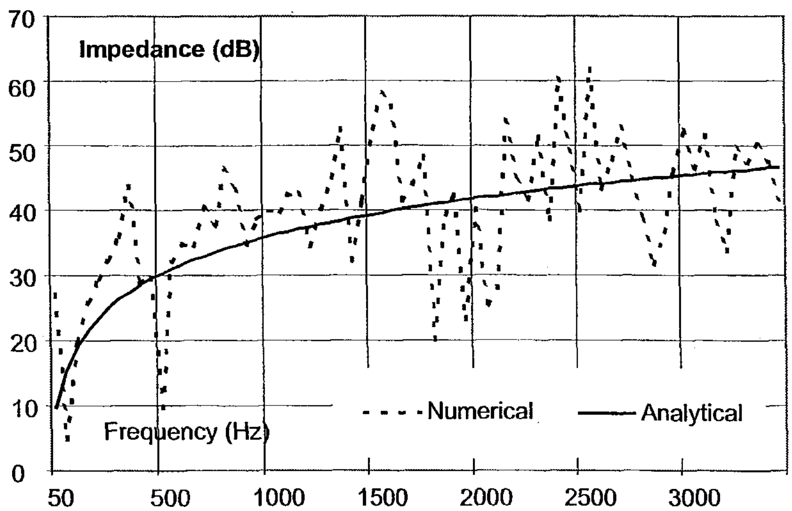

Fig.6 Radiation Impedance

6.CONCLUSIONS

A considerable amount of information has been collected from the numerical and experimental point of view, in a way to create a validated model of the muffler-cavity acoustic system. In this study numerical-experimental comparison could be considered valid up to 2600Hz (the validity of numerical model).

In this frequency range the use of radiation impedance at design stage has been proved giving a good approximation, but at some frequencies a model that considers the presence of the cavity is preferable. Results show how the numerical model for muffler design has been validated and refined.

7.ACKNOLEDGEMENTS

Authors would like to thank Mr.Lelio Citron for his precious help at set-up preparation and the carrying-out of tests.

8.REFERENCES

[1] SYSNOISE "Theoretical manual 5.4" LMS Numerical Technologies N.V. Leuven, Belgium

[2] M. Bucciarelli, A. Faraon, F.M. Giusto; Electrolux Compressor, Italy; "*An approach to Evaluate Acoustical Characteristics of Silencers Used in Hermetic Compressors for Household refrigeration*" Purdue Compressor Conference, 1994

[3] J.F. Hamilton "Measurements and Control of Compressor Noise", Purdue University, 1988, USA

[4] "Running Mode Manual 3.5B" LMS Leuven, Belgium

[5] L.E. Kinsler "Fundamental of Acoustics" John Wiley & Sons 1982